FEASIBILITY AND DESIGN STUDY OF A DRIVE BOX FOR APPLICATION IN A TANDEM REAR AXLE LIGHT TRUCK

A Thesis

Presented in Partial Fulfillment of the Requirements for the Degree Bachelor of Science with Distinction in Mechanical Engineering

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ABSTRACT

Significant improvements in load handling can be made to light trucks with the application of a tandem rear suspension. Most current concepts involve two complete axle assemblies with all the associated drive components. This concept is good, but an alternative method of transferring power to the rear most axle that is discussed in this paper provides additional benefits. This research effort resulted in a preliminary design for a drive box that will allow the driver to disengage the rear most axle when extra traction is not required. By disengaging part of the drive-line, there is an increase in system efficiency versus the current methods. The resulting design is mechanically sound, but as will be discussed, may not be an economically and weight feasible idea.

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TABLE OF CONTENTS

LIST OF FIGURES AND TABLES

1. Introduction

1.1 Motivation

In a time where light trucks are more popular to drive than ever, extensive research is going into improving them. These light trucks have progressed to a point of great comfort, convenience, and performance when compared to the pickups of yesteryear. There is, however, a point where the capability of these trucks does not meet the requirements of users who want to haul or pull large loads on occasion. These drivers have to purchase larger trucks that don't have all the comfort, are bigger and more difficult to maneuver, and are far less efficient to operate, especially at the times where the size of the truck is much larger than needed for a small or even the lack of payload. The popular option currently is the use of dual tires on each side of the truck, but this only allows for limited amount of gain in payload capacity.

A different solution to this problem could be in the form of tandem rear suspension/axles, in some instances referred to as 6x6. Many benefits come from this tire arrangement where there are two steering tires at the front of the vehicle and four drive tires at the rear, two on either side, one right in front of the other, for a total of three axles. Figure 1 shows a concept vehicle developed by Dodge that employed the concept of tandem rear axles. The tandem rear axle arrangement allows for a larger payload and towing capacity, while maintaining the size of a light duty pickup. In addition to the payload/towing capability, another benefit of the extra rear axle is improved stability and increased braking [1]. If power is supplied to the rear tires, improved traction also results. A draw-back of the tandem set-up is the added weight from an extra axle and additional parts that will wear and/or fail, requiring replacement.

Figure 1: Dodge T-Rex Concept Vehicle with Tandem Rear Axles [2]

The popular method for the tandem set-up in light trucks is to have a differential power divider between the two tires of the middle axle, and either to have an output shaft continue through the differential or to have a gear box attached to the front of the differential with an output shaft above the differential housing. The output shaft in either case is connected to a second differential power divider between the two tires of the rear most axle via a short drive shaft. This set-up is also the same that is used on nearly all large trucks and semi-tractor trucks. In an effort to make an improvement over the existing designs, the option using one main differential power divider to split power between sides with the tires on each side being connected independent of the tires on the opposite side. Even further refinement leads to a clutching system that allows only one of the two rear axles to receive power, reducing number of parts rotating during light use, thus reducing rotating inertia. Preliminary thinking was that design may offer benefits in weight savings, total system efficiency, reduced component wear, and smaller overall size.

1.2 Literature Survey – Patents

In order to get a feel of what type of similar ideas to the one of this project, a patent search was performed. Over 50 specific patents were looked at and hundreds of titles were scanned. While no patents were found that specifically described the design of this project, there were a few that were similar in certain aspects that sparked ideas.

A couple patents, more on the historical aspect of tandem axle, are pat. no. 721,705 by C. W. Hunt and pat. no. 967,728 by J. J. Charley. Patent number 721,705 is dated Mar. 3, 1903 and is the earliest application of tandem drive axle design discovered by this search. It's a primitive idea using worm gears mounted to the central drive shaft. Each axle has it own "differential." Patent number 967,728, dated 1910, is a more developed idea with a single transverse shaft, with power divided between the fore and aft rear axles via a series of bevel gears and shafts.

Patent number 2,659,246 by E. F. Norelius is the next patent of interest. Shown in Figure 2, it has a single transverse shaft that supplies power to an epicyclic gear train in the hub of one of the wheels. This epicyclic gear train acts as a differential, splitting the power between the fore and aft axles. There are various arrangements of the sun gear, carrier, and ring gear shown. Power is transferred from one axle to the other by means of a ring and pinion assembly on each axle, connected by a short splined shaft. Using ring and pinion assemblies connected via a shaft is one idea that was of interest for further investigation by this project. The epicyclic gear train idea in the tire hub is another good idea, but due to its complexity and size constraints, it may not easily be applied to this design.

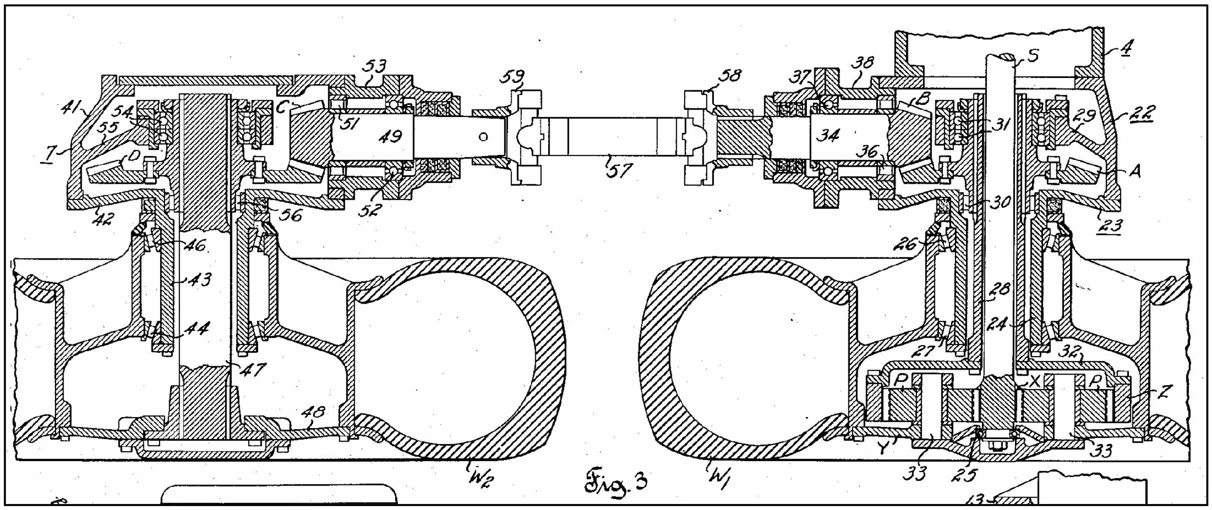

Figure 2: Figure from Patent no. 2,659,246

Patent number 3,976,154 by R. B. Clark and H. R. Anderson also uses the idea of a single transverse shaft. It to is using an epicyclic gear train, but it is located in between the two axles, not on one of the axles as in pat. no. 2,659,246. It then uses chains to transfer power from the output of the epicyclic gear train to each of the tires. While the location of the power splitter is in a different location from that of the idea for this project, the idea of using chain to transfer power from one axle to the other is similar. Also, some modified version of the epicyclic gear train may be adapted.

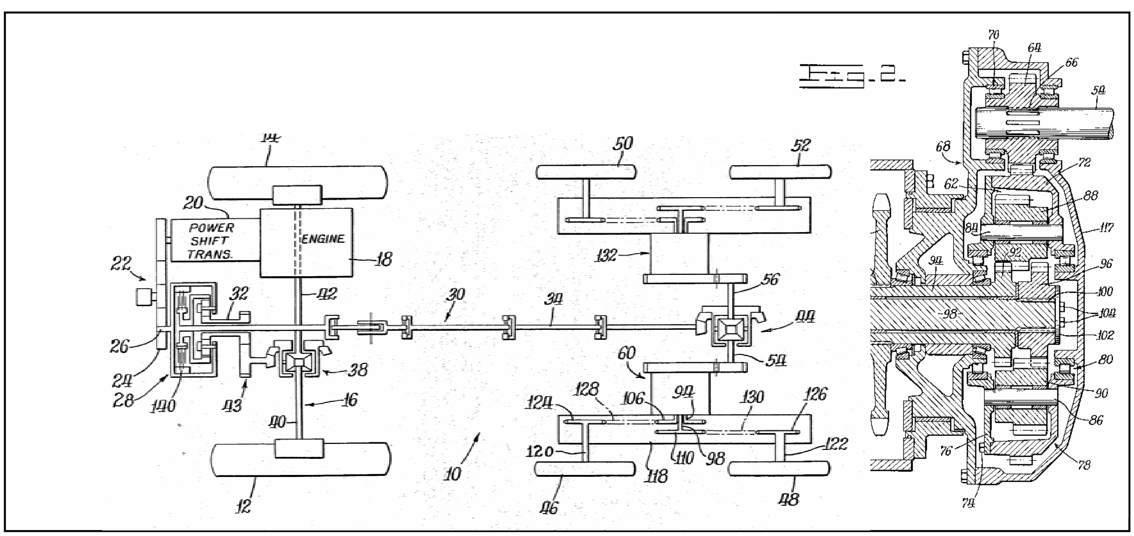

Figure 3: Figures from Patent no. 3,976,154

Patent number 4,577,711 by C. O. Butler is possibly the most similar to the idea of this project. Figures from this patent are shown in . It is a design most likely for light trucks, as is this project. It has an axle similar to production pickups with a gear wheel mounted inboard of the wheel on each side of the vehicle. This gear wheel powers a belt which drives a clutch unit mounted between the fore and aft tire. The clutch describe is a sliding pin assembly, but reference to other clutch designs such as friction plates is mentioned. The output of the clutch is then connect to a gear wheel mounted inboard of the other wheel. As stated before, this patent is very like the idea of this project, but a goal of my design is to have none of the components used for driving the second axle in motion when a clutch is disengaged. The design of patent number 4,577,711 has the first drive belt and half of the clutch unit always being in motion. One further difference between this patent and the proposed design of this research is that this patent utilizes a solid/walking beam axle design, where the proposed design is aimed primarily toward independent suspension.

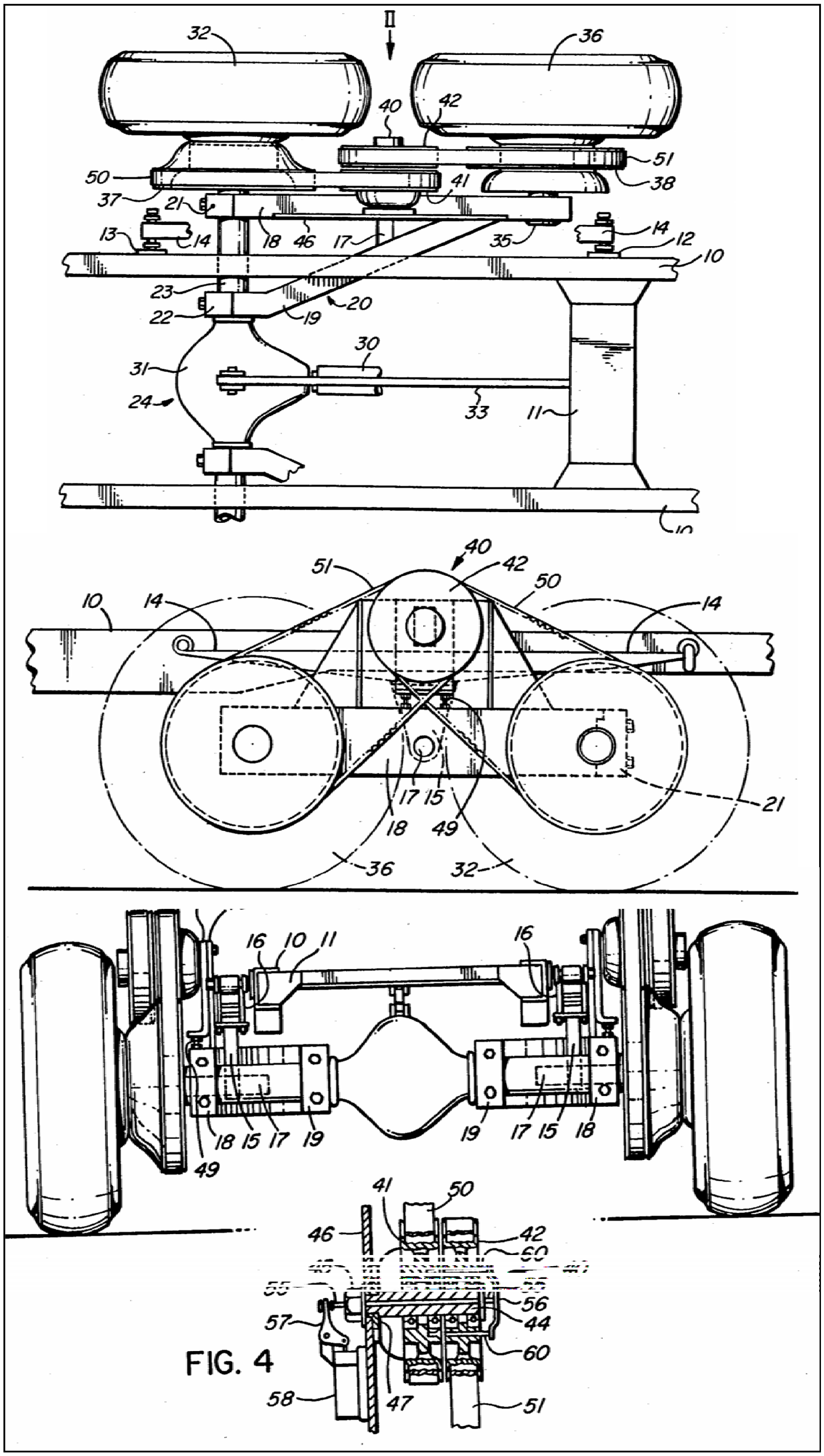

Figure 4: Figures from Patent no. 4,577,711

Patent number 6,752,235 by D. K. Bell and J. Sandler is the last patent of interest to this project. While no description is made of the type of drive system in a "rigid, substantially box-like member," it does use the idea of tandem rear axles with independent suspension. It also has one input into this "box-like member" as does my idea. A figure from this patent is shown in

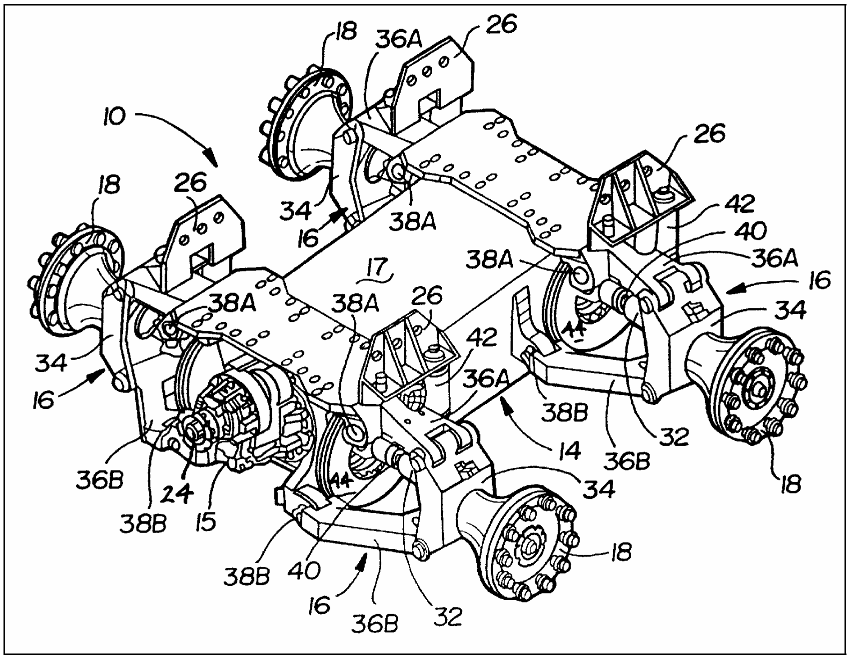

Figure 5: Figures from Patent no. 6,752,235

1.3 Design Goals

In order to make the design promising for future further development, a few goals for the design were set. The goals and objectives were set using the design process of Dym and Little [3]. Figure 6 shows the objective tree that created to identify the key objectives. With objectives at hand, a Pairwise Comparison Chart (Table 1) was constructed to rand the objectives. The highest ranked goals include being durable, compact, and cost effective. With these objectives, the drive boxes will allow the truck to also be cost effective and efficient. With proper durability, the drive boxes will not be the "weak link" in the overall drive system, thus keeping the potential for downtime of broken components to a minimum. Other design parameters taken into consideration were maintenance requirements and ease and smoothness of operation.

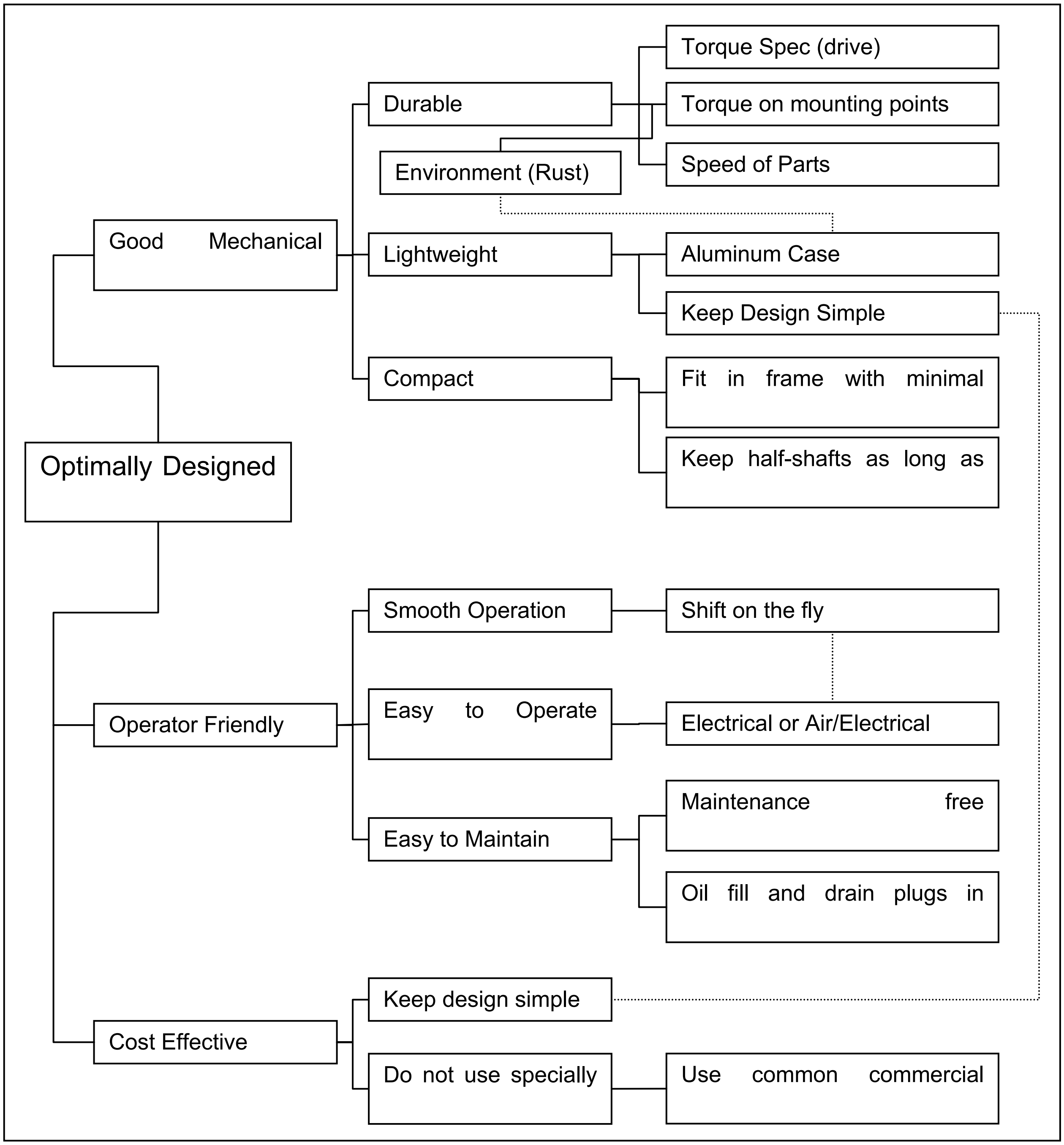

Figure 6: Objective Tree Used to Identify Key Objectives

Table 1: Pairwise Comparison Chart

Goals	Durability	Lightweight	Compact	Smooth Operation	Easy to Operate	Easy to Maintain	Cost	Score
Durability	x	1	1	1	1	1	1	6
Lightweight	0	x	0	0	1	0	0	1
Compact	0	1	x	1	1	1	1	5
Smooth Operation	0	1	0	x	1	1	0	3
Easy to Operate	0	0	0	0	x	0	0	0
Easy to Maintain	0	1	0	0	1	x	0	2
Cost	0	1	0	1	1	1	x	4

Table 2: Final Ranking from Pairwise Comparison Chart

Final Ranking	
Durability	More Important
Compact	
Cost	
Smooth Operation	
Easy to Maintain	
Lightweight	
Easy to Operate	Less Important

1.4 Thesis Content

The focus of research was to perform a feasibility and design study of a system that will transmit power to either one or both tires in tandem arrangement from one output shaft of the central differential. This work covers all the work up to the point of a preliminary design including all the pre-design work. This chapter, Chapter 1, goes into detail of the problem definition process. The design specifics are covered in chapter 2. Subsections of chapter 2 include size restraints, torque calculations, power transfer alternatives, clutch design alternatives, bearing life calculations, and key analysis. Chapter 3 discusses the final preliminary design. A conclusion based on this research is given in chapter 4.

2. Design Specifics

2.1 Size Restraints

Before many calculations were performed on the strength of components and such, it was determined that the size restraints should be set, as this would likely be one eliminating factors on design choices. As was determined in the pairwise comparison chart for ranking objectives, compactness was 2nd most important. This can be seen in the findings of the investigation for size restraints. Front suspension, CV shafts, rear axles and frames were looked at of several different trucks. All trucks were 4x4 models with single rear wheels. Table 3 includes all trucks that were examined and measured during the investigation.

Table 3: Trucks that were examined and measured

Year	Make	Model
2004	Chevrolet	Silverado 1500
2004	Chevrolet	Silverado 2500HD
2004	Dodge	Ram 1500
2004	Dodge	Ram 2500HD
2000	Ford	F-150
2005	Ford	F-150
2004	Ford	F-350
2004	Toyota	Tundra

Table 4: Dimensions measured on each truck

Rear Dimensions	Front Dimensions
Frame Width-inside to inside	Lower A-arm length
Frame Width-outside to outside	Upper A-arm length
Track Width	CV Shaft length-center of joint to center of joint
Width of Differential Housing	Width of cross member at lower A-arm mounting points
Distance between Axle Centerline and Frame Bottom	Width of cross member at upper A-arm mounting points
	Distance between CV joint mounting points

All measurements were done with a standard 25 foot tape measure. In some instances, the measurement had to be “eye-balled”, but figures should be close enough to serve their purpose. The primary reason for measuring the above dimensions was to get a feel of how big each drive box can be. Because this project is only a preliminary design to investigate feasibility for any light truck, exact dimensions were not required.

By combining the distance between CV housing mounting points and the inner width of the frame, it appears that the width of the assembly consisting of both drive boxes and the center differential should be 30 inches or less. Typical differential housings are around 18 to 20 inches wide. This corresponds to each drive box being no wider than 5 or 6 inches. Further investigation revealed that the Ford “nine inch” rear axle has a differential housing that is commonly narrowed down to 12 ¾ inches for use in drag racing and rock crawling vehicles. Little, USA6X6 [1], also uses these differentials in his conversions to tandem independent rear suspension, although in a slightly different was. The narrower width allows for a box width of 8 ½ inches, providing much needed space for packaging.

One design difference between different makes/models is the way the CV housings are mounted. The Ford trucks use a CV housing that bolts to a flange that is attached to the axle shaft. The Dodge and Toyota trucks use a CV housing that slides on splined axle shafts. The Ford mounting method is shorter, thus the axle housing can be wider. The bolt on housings is most likely the way that the CV housings will be attached to my design.

The height of the box, above axle center line can be up to 11”. Going 11” below the axle center line will likely cause ground clearance issues. For this reason, the height of the drive boxes will be kept to around 12” or less if ½ is below the axle and ½ above. It will be ok, if required, to make the box taller, only if the addition is to above the axle.

The distance between axle centerlines was suggested by Little [4] to be the diameter of the tire plus 3 inches for trucks used primarily on road use and plus 5 inches for trucks seeing a significant amount of off-road use. Basing calculations on 31 inch tires, a spacing of 35 inches was chosen.

2.2 Torque Calculations

To calculate the actual torque through the axles, a simplifying assumption of 100% power transfer was made. Then the torque from horsepower Equation 1 was used.

Equation 1: Torque from Horsepower

$$T = \frac{HP * 5252}{\left(\frac{MPH * 5280}{\left(60 * (\pi * TireDiameter) \middle/ 12 \right)} \right)}$$

This provided a graph as in Figure 7. This method, however, made an assumption that peak torque is supplied at any engine speed, an assumption that is not true especially where the vehicle is moving very slow with the engine near idle speed.

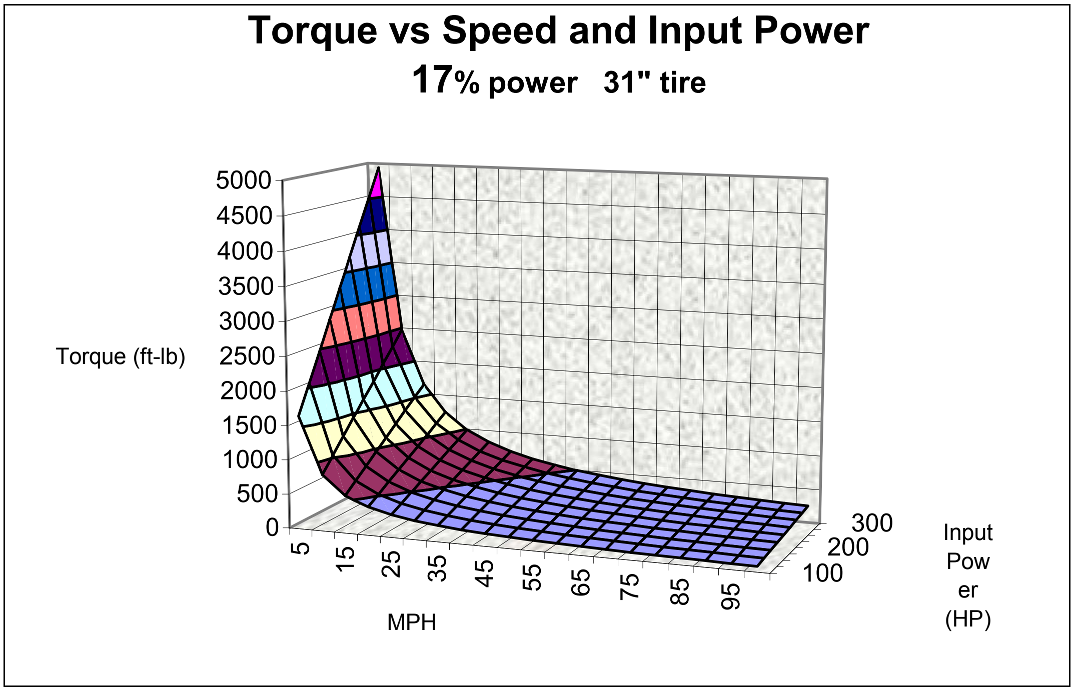

Figure 7: Torque vs. Speed and Input Power

A revised method was used that started with the peak engine torque and multiplied it by the reductions in the drive-train. This is shown in Table 5.

Table 5: Torque Transfer through Drivetrain

Case	Engine Torque (ft-lbs)	Reduction in transmission	Reduction in Transfer Case	Reduction in Rear Axle	Total torque on axles (ft-lbs)	Torque is ideally split between 3 axles	approx speed of vehicle
Extreme	420	3.1	1.98	4.56	11755	3918	5
high range, low gear	420	3.1	1	4.56	5937	1979	10
high range, mid gear	420	1.5	1	4.56	2873	958	20
high range, high gear	420	0.7	1	4.56	1341	447	50
	* 420 ft-lb is based on Cummins ISB-3.9			~170 hp			
	approx speed based on 1600 rpm peak torque						

The engine torque used is based on an engine that is planned to be used if a vehicle was constructed using the design resulting from this project. The engine torque is moderate to high torque compared to other engines. Obviously bigger diesel engines and some big gasoline engines could exceed this rating, but this value was used as a preliminary starting point.

In the extreme case where the transmission is in first gear, the transfer case is in low range, and the engine is at peak torque, the total torque ideally going through each tire is approximately 2000 ft-lbs. When looking at the vehicle in high range and first gear, the torque going through the output of each drive box is approximately 1000 ft-lbs.

Based mostly on an industry cut off that seemed to appear during investigation, a torque value that each rear tire will transmit was set to 1200 lb-ft.

2.3 Power Transfer Design Alternatives

Several ideas for power transmission have been investigated. These include inverted-tooth (silent) chain, toothed timing belt, v-belts, shafts, and roller chain. All initial sizing calculations were based on a sprocket running same speed as axle shaft.

2.3.1 Inverted-tooth (silent) chain

Using a product catalog from **Ramsey Products Corporation** [5], initial sizing of a silent chain was determined. Because of the small rotation speed of the axle shaft, a relatively large chain has to be used. Initially, a *RamPower*® ¾" pitch, 3" wide chain was chosen. This chain weighs 3.3 lbs/ft. Standard sized sprockets include 23, 25, and 38 teeth, among others. If a 23 tooth drive sprocket is used, this system would be able to transmit approximately 750 ft-lbs of torque. If 25 or 38 tooth drive sprockets were used, torque would increase to 825 ft-lb and 1200 ft-lb respectively. Weights of sprockets and chain for the 23, 25, and 38 tooth systems are 75 lb, 84 lb, and 110 lb, respectively. These weights are largely dependent on how the sprockets are used, i.e. how much they get bored out. Approximate cost for 2 sprockets and 1 chain is $1160.

2.3.2 Belts – Toothed timing and V-belt

A computer software program from **Gates Corporation** [6] was used for preliminary sizing belts. The toothed timing belt system investigation resulted in the following. Using a 55 mm wide, size 14MGT belt on 52 tooth pulleys (~9.12" dia), approximately 500 ft-lb of torque could be transmitted. A much larger diameter pulley/sprocket is needed for the belts because they are not as strong as a chain in tension. This causes the weight and rotating inertia to increase as well. Largely dependent on how the sprockets get mounted; preliminary weights show the belt and two pulleys weighing approximately 75 lb, costing approximately $1200.

The V-belt system also has the same problem with tensile strength, requiring large pulleys. Several variations were looked at, ranging from 5 to 10 rib belts running on pulleys around 10” diameters. Weights of V-belt systems compared to toothed timing belt systems were 50%-100% heavier because of increased width and diameter. Cost was slightly lower at approximately $1000. Both belt systems proved to be unviable options due to torque limitations.

2.3.3 Shaft

Another idea investigated is using a 90^o hypoid gear setup, or a ring and pinion. Initial thinking was to use a removable third member differential off of a small SUV. At first thought it didn’t make much sense to use differential assemblies on each side of the vehicle in addition to the main center one, when the original idea was to eliminate the use of one of two differentials. Torque ratings were not calculated, but such arrangements are used to transmit all the power from an engine to the ground on all types of vehicles. Weight was estimated to be in the range of 90 lbs. One advantage of this method is that if a mass manufactured third member were used, it may be simpler and cheaper to make the drive boxes of this project.

2.3.4 Roller Chain

Roller chain was also investigated using the **Diamond Chain Selection Software** [7]. To transmit 750 ft-lbs at 100 rpm (14 HP), a double strand #80 chain could be used with 27 tooth sprockets. To transmit 1200 ft-lbs at 100 rpm (22 HP), a double strand #100 could be used with 28 tooth sprockets. The #80 system uses sprockets that are ~8.5" while the #100 system uses sprockets that are ~ 12" in diameter. The cost was approximately $800 for 2 sprockets and 1 chain with a weight of about 145 lbs. Large sprocket diameters and excessive weight led to the elimination of this option.

2.3.5 Gear Train Series

One final option that was investigated for power transmission was using a spur gear train series of 5 gears. In addition to likely being inefficient and extremely heavy, this design proved to be unfeasible by preliminary calculations.

2.3.6 Summary

Based on above data, either of the belt options doesn't seem viable. The roller chain option may work, but the sprocket size is getting fairly large, making packaging in the already defined area difficult. A shaft/hypoid gear system and an inverted-tooth chain system still appear to be viable options. The initial conclusion was that the gear/shaft seemed to be the best fit based on mechanical, weight, and cost aspects. Once solid CAD modeling began, the hypoid gear system had to be eliminated because of width constraints. The inverted chain was the final chosen method for power transmission.

As stated above, all systems were analyzed with the input sprocket/pulley turning at the axle speed. This is the major down fall for the systems, because as shaft speed decreases with a constant input power, i.e. shifting vehicle into low gear, torque increases. A few ideas have been contemplated with using a gear system to increase the speed of the input sprocket. This would allow smaller components to be used, but would increase the complexity and likely the cost of the design.

Another aspect of the design that wasn't discussed is the center distance between shafts. This distance will be 35 inches. While this is a relatively long distance for silent chain drive, it is still with in the 60 pitches max recommendation if ¾" pitch is used. Durability and service issues shouldn't be of concern because the inverted chain is the method used in a majority of 4x4 transfer cases which have proven records in these areas.

2.4 Clutch Design Alternatives

As stated in the introduction, having a clutch mechanism capable of disengaging the rear most axle can be beneficial by reducing the number of rotating parts when the truck is in light use. The following is a overview of possible clutching mechanisms investigated.

2.4.1 Electromagnetic Clutch – Single Surface

One of the first clutches looked at was a Warner Electric PC-1225 electric clutch [8]. This is an industrial DC electromagnetic clutch. A combination of factors quickly eliminated such a design. The maximum static torque of this design is 465 lb.ft., while its outside diameter is just over 13 inches. The torque value is only 1/3 of the desired value of 1200 lb.ft. The 13 inch height is also larger than the size restraints set.

2.4.2 Electromagnetic Clutch – Multiple Disc

While similar in concept, a design difference of using multiple discs within an electromagnetic clutch such as the model EMA-0950 by The Carlyle Johnson Machine Company [9] creates a clutch that is more compact and able to transmit higher amounts of torque. This clutch is just over 10 inches in diameter yet able to transmit 1200 lb.ft. of static torque. The clutch comes in two halves, with the coil half supported on shaft with a set of ball bearing and the drive cup half mounted to the part to be driven, in this case the sprocket. The two halves are separated by a series of inner and outer discs. When electric power is supplied to the unit and the coil becomes energized, the clutch pack is pulled and held together by magnetic flux passing through it. Power is transmitted via frictional effects between the discs. Between each disc is a separator ring which serves to positively disengage the clutch, reducing the drag torque in the disengaged position. [9] The unit can run dry or in oil, so lubrication of power transmission components will not affect the performance of the clutch.

2.4.3 Toothed Clutch

An option that is very strong and compact is a toothed clutch. An example of such a clutch is as shown in Figure 8, made by Dayton Superior Products [10]. While the principle of operation is very simple, the design poses a few drawbacks. The first is that smooth engagement is unlikely. The tooth design that allows for high amounts of torque causes positive stop points that will result in jerky engagement. Additionally, the clutch requires some sort of shifter mechanism such as a hydraulic or pneumatic actuator. Incorporating such a part into the design only further complicates the overall design. Packaging of the actuator also comes of big concern.

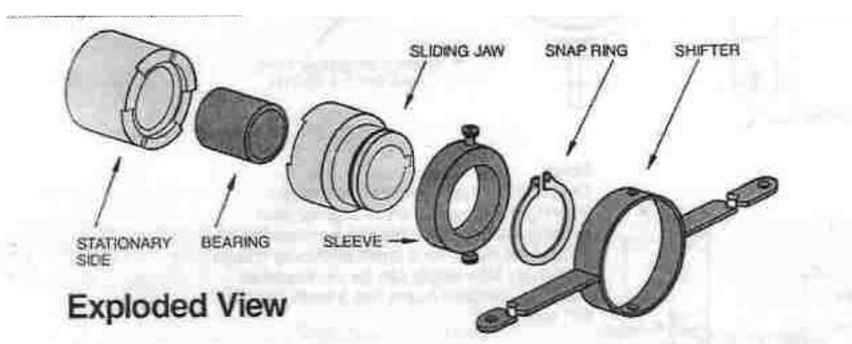

Figure 8: Toothed Clutch

2.4.4 Pneumatically Actuated Clutch

Another alternative that was briefly looked at was using a pneumatic cylinder to supply the pressure to engage either a disc pack or a cone clutch. Preliminary calculations showed that the cone clutch was infeasible, while the disc pack clutch may be. Further investigation revealed that such a specialized design was pulling away from one of the objectives of using commercially available parts. Also as with the toothed clutch, packaging of the actuator could have posed problems.

2.4.5 Summary

Based on the discussion above, the multiple disc electromagnetic clutch was chosen as the method for disengaging the drive mechanism for the aft axle.

2.5 Bearing Life Calculations

Several ball bearings are used in the design. To ensure durability of the entire design, life analysis on the bearings was performed. The first thing that was to be determined was what the life of the bearing should be. It was estimated that the life of the bearings should be 150,000 miles. Different loading on the bearings when engaged versus disengaged were split up into 9,000 miles engaged (6%) and 141,000 miles disengaged. Based on the size and weight of the sprocket and torque being transmitted, a loaded force of 3180 lb/2 bearings and an unloaded force of 48 lb/2 bearings was determined. Using equations 11-3 and 11-4 from Collins [11] resulted in a C_d,req of 5645 lb. The bearing that was selected was a single row ball bearing, #6208. This bearing has a C_d of 6900 lb, over that of the required amount.

2.6 Keyway Analysis

Another small, but significant portion of the design is the use of several keyways. Calculations were performed on both the shaft and the key. The shaft was of concern because the keyway acts as a stress concentrator, in effect making the shaft weaker. The fatigue stress on the shaft was found to be 74 ksi. A shaft of 4340 has a fatigue strength of 100 ksi, well in excess of the stress level. The normal stress on the key was found to be 30 ksi. A low carbon steel such as 1020 has a fatigue limit in excess of this at 33 ksi. Using a material that provides a factor of safety just over 1 allows the key to act as a fuse in the system. The small cost of the key replacement is far less than a clutch or chain replacement if one of the components were to malfunction and jam the system.

3. **Final Preliminary Design**

Once all the design options were looked at and evaluated, a preliminary design was modeled in Solidworks® CAD software. The final design incorporated an inverted chain and an electromagnetic multiple disc clutch. An aluminum box was designed with mounting brackets. The box was made narrower at the back to reduce weight and to increase CV shaft length of the rear axle to allow for more articulation. The differential chosen is a modified Ford 9" rear end, with an overall width 12.75 inches. The modification to narrow the axle housing allowed the overall width to just over 30 inches. The weight of the each box is ~175 lb.

Mounting points for CV housings are shown just to the outside of the box in Figure 9. The attached file on the OSU Knowledge Bank Website, TandemDriveBox.exe is a self executable file of the solid model that can be rotated and zoomed in on for further investigation.

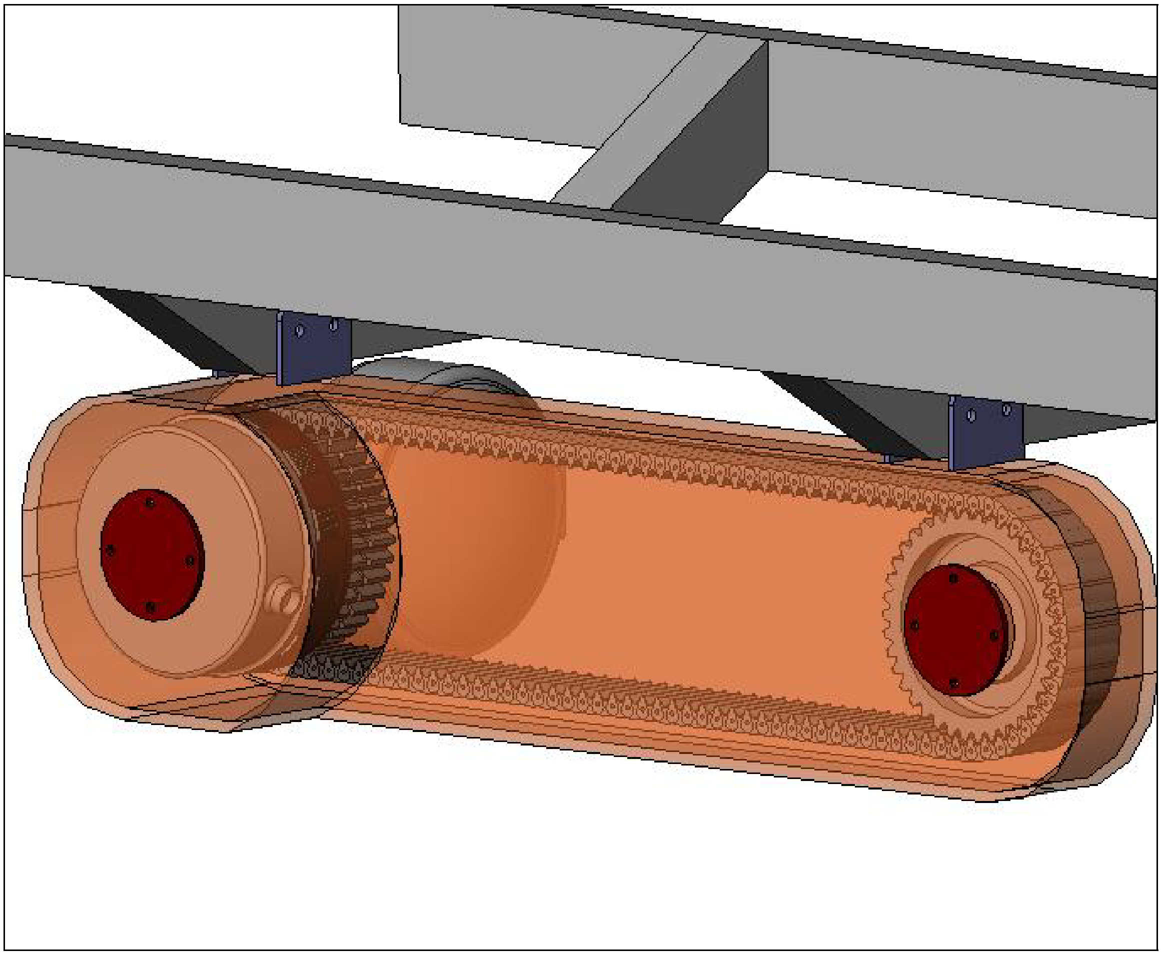

Figure 9: View of Solid Model of Final Preliminary Design

4. Conclusion

A preliminary design was arrived at that is mechanically sound and a viable solution to building a pickup with tandem rear axles. However, it should be noted that it is not a very feasible or logical solution. Higher weights and cost versus other designs are the primary reasons. The primary reason the design did not turn out as beneficial as originally thought is due to the under estimation of the effect that the relatively slow speed of the components would have on the size of the components required.

A possible solution for future investigation is using the rear axle as the means for hybrid electric power. While still a rough concept and beyond the scope of this project, having the rear axle connected to the other axles only through road contact and driven by an electric motor would provide additional benefits of improved power and fuel economy.

References

[1] Little, Daniel B., 6x6 Conversions, 12/2003, USA6x6, 2/20/2004, <http://www.usa6x6.com/6x6conversions/6x6concepts/>.

[2] T-Rex, DaimlerChrysler, 3/03/2004, <http://www.dcxevents.com/>.

[3] Dym, Clive L, and Patrick Little. Engineering Design: A Project-Based Introduction, 2nd Edition. Wiley, 2004.

[4] Little, Daniel B., E-mail. 1/15/2005.

[5] Ramsey Product Corporation, 2/16/2005, <http://www.ramseychain.com>.

[6] Gates Corporation, 2/16/2005, <http://www.gates.com>.

[7] Diamond Chain Company, 2/16/2005, <http:www.diamondchain.com>.

[8] Warner Electric. PC-1225 Normal Duty Clutch. 2005.

[9] The Carlyle Johnson Machine Company, LLC. Electric Multiple Disc, Clutches and Brakes, Model EMA. 2005.

[10] Dayton Superior Products Co., Inc. 3/7/2005, <http://www.daytonsuperior products.com/tooth.htm>.

[11] Collins, J. A., Mechanical Design of Machine Elements and Machines: A Failure Prevention Perspective. Wiley, 2002.